AAA-RPO-TRNS-01-0021
RPO-R00-O-TNS-000103
Revision 0
March 2000

Advanced Accelerator Applications
Project Technical Note
Research Project Office

Heat Exchanger Analysis for the LBE Material Test Loop

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Document Number:
AAA-RPO-TRNS-01-0021
RPO-R00-O-TNS-000103
Revision 0

Category: 1

Abstract:

This document summarizes the calculations that were performed to determine the heat exchanger heat transfer capabilities, pressure drop, and internal pressure.

Author:

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Approval:

Project Leader	**Signature on File – 9/4/2001**
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HEAT EXCHANGER ANALYSIS COMMENT RESOLUTION SHEET

COMMENT	RESOLUTION	Initials
1. First page of main text describes cooling water flowing downward in the annulus and back up the center hole. The drawing shows the opposite direction.	Corrected.	DJW
2. In the next to last line on the first page, the outer radius is shown as 27.2 mm. For a 2 in. Sch 40 pipe the inner radius is 26.2 mm.	Corrected.	DJW
3. On the second page, the heat exchanger power is given as 55 kW. I calculated that the LBE temperatures and flow rate give a power of 52.7 kW. I also calculated that the water temperatures and flow rate give a power of 52.0 kW. This would reduce the discrepancy somewhat between the calculated heat exchanger power and the Russian numbers described in the second paragraph on p. 2.	Corrected.	DJW
4. In the upper table on p. 3, the third entry should be changed from 0.56 to 0.59 based on the change noted for Rth6 in the Check of Russian Design Point Calculations in Appendix A.	Corrected.	DJW
5. In the lower table on p.3, for 1.5 m/s in 1" Sch. 40 pipe I get a flow rate of 8.36E-4 m3/s vs the 7.75E-4 m3/s given there. This is based on an ID of 1.049 in. or 2.664 cm.	Corrected.	DJW
6. On page 4, at the start of the first sentence in the second paragraph under Pressure Drop Analysis, there is a missing "on" after "The resulting pressure drop."	Corrected.	DJW
7. Further down in the same paragraph on p. 4, the pressure drop in the water system is 36 psi rather than 26 psi based on changes noted in Appendix B.	Corrected.	DJW
8. On p. 5, the following changes occur because of changes noted in Appendix C. The volume ratio changes from 1.28 to 1.39. The gas space nominal pressure increases from 50 to 54 psi and the maximum pressure including the safety factor increases from 75 to 81 psi.	Corrected.	DJW

9. To check Ap. A, I wrote a separate spread sheet and did the analyses to check the nominal case, the case with the 5.67 Dryer Nu number, the case with a 12.5 Nu number, and the case with the length increased to 114 cm from the nominal 76 cm. All the results were very close to what you got. The only change I noted was in the calculation of the RTH6, the conductivity of lead bismuth is given as at 150 C but the value used corresponds to the 200 C value in your table. (11.73 vs 11.15 W/m-C) I didn't check the cases you did for the range of flow conditions.	Corrected.	DJW
10. On the second page of Appendix B, the flow coefficient should be 31 rather than 0.31 as we discussed earlier.	Corrected.	DJW
11. On p. 2 of Ap. B, the volume flow rate for the flow meter is given as 11 gpm, which is 6.94E-4 m3/s rather than the 6.94E-3 m3/s that was used. This cases the velocity to be 2.44 m/s rather than 24.4 m/s and the K to be 8.34 rather than 0.083.	Corrected.	DJW
12. When the change given in number 11 is put into the equations on page 3 of Ap. B, the pressure drop increases from 25.9 psi to 36.0 psi.	Corrected.	DJW
13. On p. 1 of Ap. C, the volume of the cylinder at the lower right corner is 183.3 cm3 if you use the 3.12 cm radius rather than the 3.10 cm.	Corrected.	DJW
14. On p. 2 of Ap. C, four lines up from the bottom the last term in parentheses should be 1.66 rather than 1.86. That decreases the result from 3.14 to 2.51 cm3. This impacts the last 3 lines on the page to give a V3 of 301.2 cm3 and a total volume of 1073 cm3 rather than 1148 cm3.	Corrected.	DJW
15. On p. 3 of Ap. C, the first term in the V1 equation is then 183.3 cm3 rather than 180 cm3 and this gives a total at the bottom of the page of 1419 cm3 rather than 1416 cm3.	Corrected.	DJW
16. On p.4 of Ap. C, the VC equation then reads 1073-183=890 (all cm3)	Corrected.	DJW

17. On p. 4 of Ap. C, the VE equation then reads 1419-183=1236 (all cm3) so the result doesn't change here.	Corrected.	[illegible]
18. On p. 4 of Ap. C, the volume ratio is then 1236/890=1.39. This gives a nominal pressure ratio of 3.67 and with safety factor a 5.5 pressure ratio for corresponding pressures of 54 and 81 psi.	Corrected.	[illegible]

Distribution List

Ammerman, Curtt	H821
APT-RMDC	C341
APT-RPO	H816
Pasamehmetoglu, Kemal	H816
Quintana, Lawrence (QA)	H809
Smith, Brian	H821
Tomei, Tony	H836
Woloshun, Keith	H855

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Heat Exchanger Analysis for the LBE Material Test Loop

Introduction

The heat exchanger is used in the Material Test Loop (MTL) for removing excess heat from the liquid lead-bismuth eutectic (LBE). The heat exchanger is a counter-flow, concentric tube design. Heat is removed from the LBE with water. The heat exchanger design was supplied by the Russians. It was modified slightly by project engineers and designers and then fabricated in the U.S.

This document summarizes the calculations that were performed to determine the heat exchanger heat transfer capabilities, pressure drop, and internal pressure.

Heat Exchanger Overview

A sectional view of the heat exchanger is shown in Fig. 1. LBE flows in the outer annulus and water flows in the innermost section. The LBE enters at the top and flows downward. The water enters and exits at the top. The water flows downward through the central tube, then upward in an annulus surrounding the central tube.

Due to the high temperature gradient from the LBE to the water, an intermediate annular cavity is located between the hot and cold sides of the heat exchanger. The base of this cavity contains a small amount of LBE with an argon cover gas. This LBE (when melted) can be forced up into the annulus creating a variable heat transfer surface area. This variable area provides variable heat transfer capability enabling the MTL facility to operate over a wide range of conditions. This intermediate annular cavity is sealed with two bellows sections that allow the central pipe structure to move vertically. A handle at the top of the heat exchanger is attached to a threaded rod that can be rotated to provide the central structure with 60 mm of vertical movement. This movement allows the small volume of LBE to be compressed and forced into the intermediate annulus.

Figure 2 shows an enlarged view of the heat exchanger cross section. The heat exchanger is constructed of 316 stainless steel pipe and tubing. The outer shell is 3 _ schedule 40 pipe with an inner radius of 45.1 mm (dimensions were obtained from the ESA-DE Heat Exchanger Assembly drawings: #142Y600924). The LBE flows in an annulus between this outer shell and the outside of a 2 _ schedule 40 section of pipe. The inner radius of this LBE annulus is 36.5 mm. The intermediate annular cavity is created between the 2 _ schedule 40 pipe and a 2 schedule 40 pipe section. The outer and inner radii of this annular cavity are 31.4 and 30 mm, respectively. A final annulus for water flow is formed between the 2 schedule 40 pipe and a section of 1 _ OD tubing with a 0.083 wall thickness. The outer and inner radii of this annulus are 26.2 and 22.2 mm, respectively. The water that flows in this annulus is returned upward through the inside of the tubing.

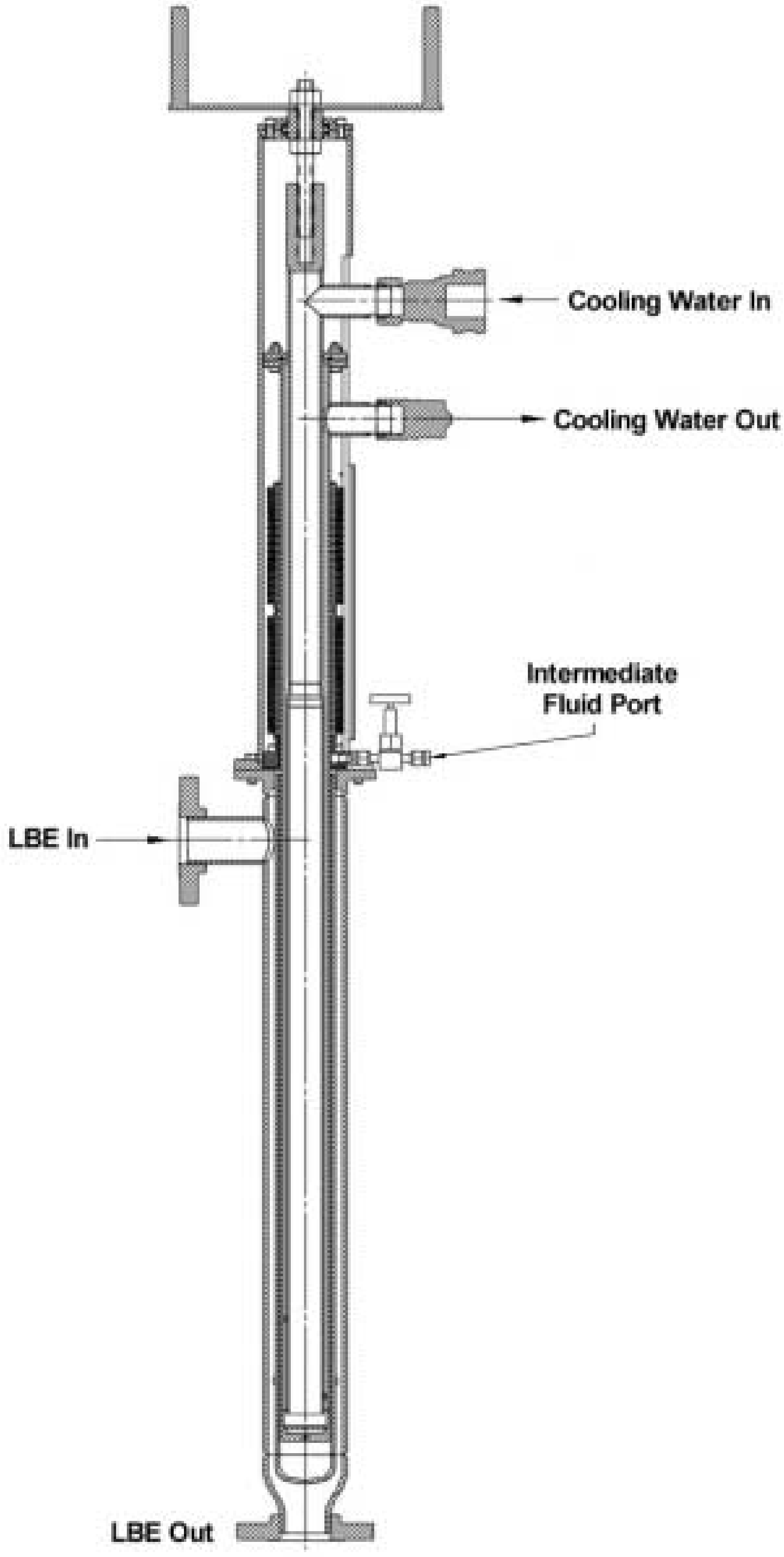
Cooling Water In
Cooling Water Out
Intermediate
Fluid Port
LBE In
LBE Out

Fig. 1. Heat Exchanger Drawing.

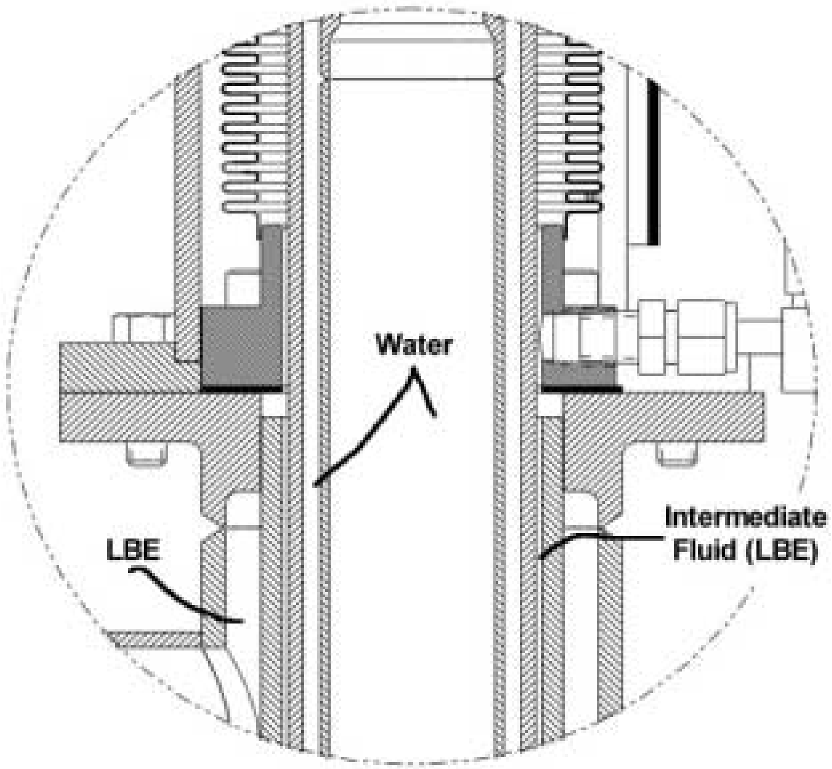

Fig. 2. Enlarged View of Heat Exchanger Cross Section.

Design Point Heat Transfer Analysis

The Russians designed this heat exchanger to the following conditions:

Parameter	Value
Heat Exchanger Power (kW)	52
LBE Inlet Temperature (¡C)	350
LBE Outlet Temperature (¡C)	250
Water Inlet Temperature (¡C)	25
Water Outlet Temperature (¡C)	40
LBE Flow Rate (m^3/hr)	1.25
Water Flow Rate (m^3/hr)	3.0

An analysis was performed to check this design point and is shown in detail in Appendix A. From the LBE to the water, there are five thermal resistances that are considered:

Convection from LBE to wall	$R_{th,8}$
Conduction through 5.16-mm-thick wall	$R_{th,7}$
Conduction through intermediate LBE layer	$R_{th,6}$
Conduction through 3.91-mm-thick wall	$R_{th,5}$
Convection from wall to water	$R_{th,4}$

This heat transfer analysis was performed using two different predictive methods to obtain the liquid metal heat transfer component. The first method employs the numerically generated tables of Kays and Leung [1]. Kays and Leung performed a theoretical study of heat transfer in annular channels for hydrodynamically developed turbulent flow. Their results, for the cases of uniform heat flux from either the inner or outer wall only, are presented in tabular form. Using Kays and Leung for the liquid metal heat transfer, the analysis of the Russian design point shows a heat transfer rate in the heat exchanger of 39.6 kW. This result is 24% less than the 52 kW predicted by the Russians.

Dwyer reports in the Sodium-NaK Engineering Handbook [2] that in the very low Peclet number range, the Kays and Leung results agree well with his own semi-empirical correlation. At higher Peclet numbers, however, the Kays and Leung results significantly underpredict Nusselt number. This assessment is confirmed by Kays and Crawford [3] who state that at very low Prandtl numbers, the results for Nusselt number may be low. The assessment of the Russian design point was performed again using the correlation of Dwyer (also shown in Appendix A). The results from this assessment show a heat transfer rate of 39.1 kW, which is nearly the same as the Kays and Leung prediction. The Peclet number in this case is 324. Apparently the Kays and Leung prediction is performing well at this Peclet number.

Both the Kays and Leung and the Dwyer methods of predicting heat transfer assume fully developed turbulent flow. Dwyer points out, however, that the thermal entry length for liquid metals could be as high as 30 diameters, or more than half of the heated length of the heat exchanger. Because of a lack of Nusselt number data for this specific case, the thermal entry length effect was estimated by doubling the liquid metal heat transfer coefficient over the entire heated length of the heat exchanger. The result of this analysis showed that the heat transfer rate only increased to 42.9 kW, which is still 18% less than the Russian prediction.

To assess the controlling resistances in the heat transfer path, the five thermal resistances computed for this analysis are normalized by the LBE convection resistance ($R_{th,8}$). These normalized resistances are shown below:

Convection from LBE to wall	1.00
Conduction through 5.16-mm-thick wall	1.69
Conduction through intermediate LBE layer	0.59

Conduction through 3.91-mm-thick wall	1.54
Convection from wall to water	0.88

All resistances are of the same order of magnitude. Large changes in the LBE convection resistance, therefore, have a small overall effect on the total heat transfer rate.

All heat transfer surfaces are assumed to be clean and free from fouling. In practice, it is likely that some surface contamination will be present, thus further reducing the efficiency of the heat exchanger.

Off-Design Heat Transfer Analysis

To provide an assessment of how the heat exchanger will perform at operating conditions other than the design point, a range of conditions was examined. Four different volume flow rates and three different LBE inlet temperature conditions were combined resulting in 12 different off-design cases. The three inlet temperatures examined are 350¡C, 450¡C, and 500¡C. The volume flow rates examined are shown below along with their corresponding sources:

3.47×10^{-4} m^3/s	Russian flow condition (1.25 m^3/hr)
8.36×10^{-4} m^3/s	1.5 m/s in 1 schedule 40 pipe
2.16×10^{-3} m^3/s	1.0 m/s in 2 schedule 40 pipe
4.33×10^{-3} m^3/s	2.0 m/s in 2 schedule 40 pipe

To simplify the analysis, LBE properties were evaluated at an estimated mean temperature of 410¡C. The water flow rate used for all calculations was the Russian design flow rate of 3.0 m^3/hr (13.2 gpm). Details of the analysis are shown in Appendix A.

The results of this off-design analysis are shown in Fig. 3. Calculations in Appendix A were performed both with the Kays & Leung and with the Dwyer predictions. Because of the reported discrepancies in the Kays & Leung prediction at low Prandtl/high Peclet numbers, the Dwyer correlation was used for the data shown in Fig. 3. As expected, the heat transfer rate increases both with increasing flow rate and inlet temperature.

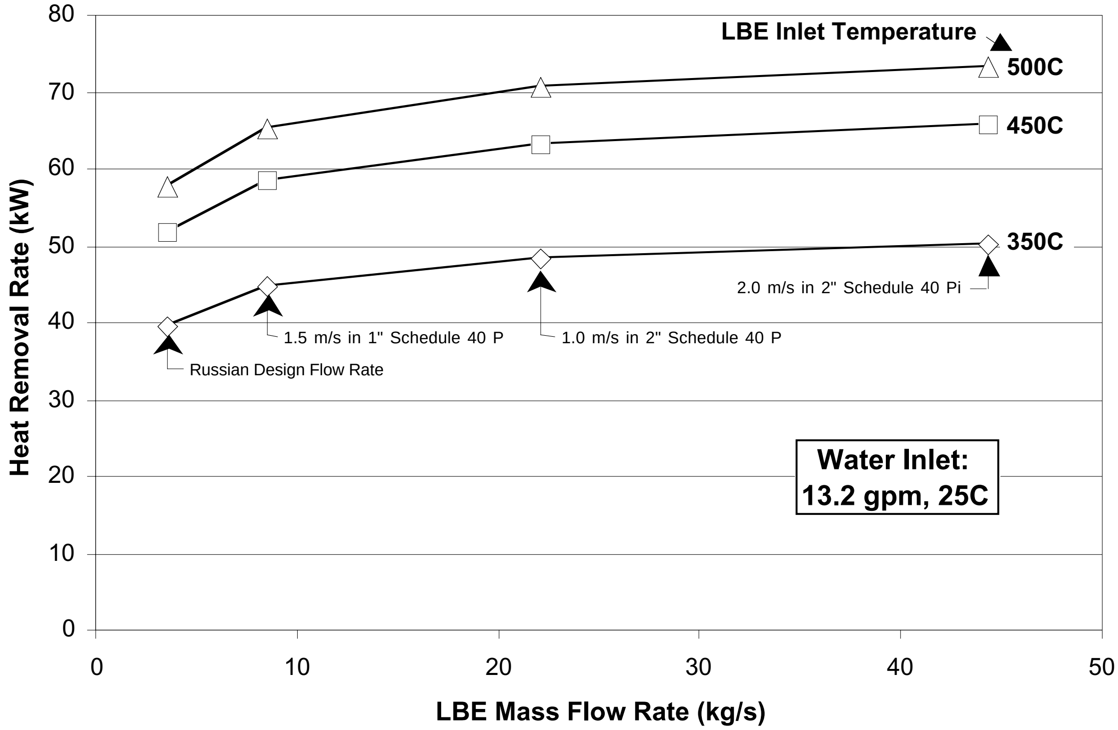

Fig. 3. Off-Design Heat Transfer Rates

Pressure Drop Analysis

Pressure drop on both the LBE and water sides of the heat exchanger was estimated. The flow conditions used for these estimates were the Russian design point flows. For the LBE side, only the pressure drop through the heat exchanger is analyzed. For the water side, external plumbing including valves, fittings, and flow meters was included in the analysis. The detailed results of these analyses are shown in Appendix B.

The resulting pressure drop on the LBE side was approximately 0.5 psi. This value is small because of the relatively large flow area and corresponding low velocity. The resulting pressure drop for the water system was approximately 36 psi. The sources used in this analysis were Idelchik [4], Swagelok [5], and Munson et al. [6].

Internal Pressure Analysis

An analysis was performed to determine the maximum pressure in the intermediate section of the heat exchanger. The pressure in this section will rise, both because of a temperature rise during operation, and because of the volume reduction during compression of the intermediate LBE. The detailed results of this analysis are shown in Appendix C.

The expanded-to-compressed volume ratio in the intermediate space is 1.39. The ratio in temperature from maximum operating to room temperature is 2.64. Using ideal gas relationships, the resulting pressure in the gas space is estimated to be 54 psi. If a safety factor of 1.5 is used, the maximum pressure expected in the gas space is 81 psi. The intermediate space was tested to 100 psig at room temperature. Room temperature hydrostatic testing was also performed on the LBE side and on the water side of the heat exchanger to pressures of 250 psig and 150 psig, respectively.

References

[1] Kays, W.M. and Leung, E.Y., Heat Transfer in Annular Passages: Hydrodynamically Developed Turbulent Flow with Arbitrarily Prescribed Heat Flux, *Int. J. Heat Mass Transfer*, (6): 537-557, 1963.

[2] Foust, O.J. (ed), *Sodium-NaK Engineering Handbook, Vol. II: Sodium Flow, Heat Transfer, Intermediate Heat Exchangers, and Steam Generators*, Gordon and Breach, New York, 1976.

[3] Kays, W.M. and Crawford, M.E., *Convective Heat and Mass Transfer, 3rd Edition*, McGraw-Hill, New York, 1993.

[4] Idelchik, I.E., *Handbook of Hydraulic Resistance, 3rd Edition*, Begell House, New York, 1996.

[5] *Swagelok Fitting and Valve Catalog*, Albuquerque Valve and Fitting Company, Albuquerque, New Mexico.

[6] Munson, B.R., Young, D.F., and Okiishi, T.H., *Fundamentals of Fluid Mechanics*, John Wiley and Sons, New York, 1990.

Appendix A
Heat Transfer Analysis of Russian Design Point and Off-Design Conditions

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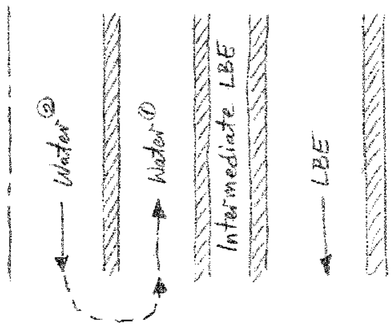
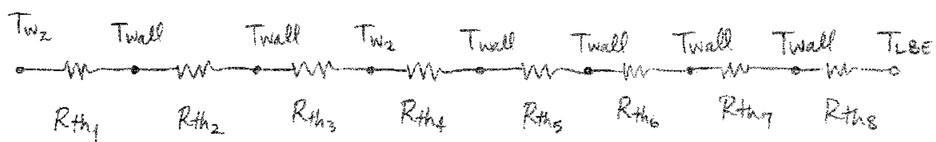
Check of Russian Design Point Calculations



	dia	rad
OD of Water②	40.23 mm	20.12
Wall thickness	(2.11 mm)	
ID of Water①	44.45 mm	22.23
OD of Water①	52.50 mm	26.25
Wall thickness	(3.91 mm)	
ID of Int. LBE	60.33 mm	30.16
OD of Int. LBE	62.71 mm	31.36
Wall thickness	(5.16 mm)	
ID of LBE	73.03 mm	36.52
OD of LBE	90.12 mm	45.06

Thermal Resistance Network



R_{th_1} – Convection of Water②

$$\dot{V} = 3\ m^3/hr = 8.33\times10^{-4}\ m^3/s$$
$$A_c = \pi/4\,(40.23\times10^{-3}\ m)^2 = 1.27\times10^{-3}\ m^2$$
$$u = \frac{\dot{V}}{A_c} = 0.65\ m/s$$

- Water properties @ 32°C ≈ 305K

$$\rho = 995\ kg/m^3$$
$$\mu = 7.69\times10^{-4}\ kg/m\,s \qquad \nu = \frac{\mu}{\rho} = 7.73\times10^{-7}\ m^2/s$$
$$k = 0.620\ W/m\,K$$
$$P_r = 5.20$$
$$C_p = 1178\ J/kg\,°C$$

$$Re_D = \frac{uD}{\nu} = \frac{(0.65\ m/s)(40.23\times10^{-3}\ m)}{7.73\times10^{-7}\ m^2/s} = 33829$$

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$Nu_D = 0.023\, Re_D^{0.8} Pr^{1/3}$ (Dittus-Boelter)

$Nu_D = (0.023)(33829)^{0.8}(5.20)^{1/3} = 167.4$

$h = Nu_D \frac{k}{D} = (167.4)\frac{(0.620\ W/mK)}{(40.23\times10^{-3}\ m)} = 2580\ \frac{W}{m^2\,°C}$

$\therefore R_{th_1} = \frac{1}{hA} = \frac{1}{h\pi DL}$ (calc. per unit length)

$R_{th_1} = \frac{1}{(2580\frac{W}{m^2°C})\pi(40.23\times10^{-3}m)} = \underline{3.067\times10^{-3}\ \frac{m°C}{W}}$

▶ R_{th_2} – Conduction through steel

$k_{ss}(400K) = 15.2\ W/mK$

$R_{th_2} = \frac{\ln(D_2/D_1)}{2\pi k_{ss} L} = \frac{\ln\left(\frac{44.45}{40.23}\right)}{2\pi(15.2\ W/mK)} = \underline{1.044\times10^{-3}\ \frac{m°C}{W}}$

▶ R_{th_3} – Convection of Water①, internal wall

$\dot{V} = 8.33\times10^{-4}\ m^3/s$

$A_c = \frac{\pi}{4}(OD^2 - ID^2) = \frac{\pi}{4}\left[(52.50\times10^{-3}m)^2 - (44.45\times10^{-3}m)^2\right] = 6.13\times10^{-4}\ m^2$

$u = \frac{\dot{V}}{A_c} = 1.36\ m/s$

$D_h = OD - ID = 8.05\times10^{-3}\ m$, $r^* = \frac{ID}{OD} = 0.847$

$Re_{D_h} = \frac{uD_h}{\nu} = \frac{(1.36\ m/s)(8.05\times10^{-3}\ m)}{(7.73\times10^{-7}\ m^2/s)} = 14163$

- Use tabular data of Kays & Leung and interpolate
  - A) uniform wall heat flux
    - $q''_{inner} \cong 0$
    - thermal entry length negligible (conservative)
    - $r^* \cong 0.8$

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- then for $r^* = 0.8$ w/ outer wall heated and inner wall insulated

Pr	$Re_D = 10^4$ Nu_{oo}	$Re_D = 14163$ Nu_{oo}	$Re_D = 3\times10^4$ Nu_{oo}
3.0	61.3		142
5.2	$\boxed{73.5}$	$\boxed{94.4}$	$\boxed{174}$
10.0	100		243

$$h = Nu_D \frac{K}{D_h} = (94.4)\frac{(0.620\ W/mK)}{(8.05\times10^{-3}\ m)} = 7270\ W/m^2{}^\circ C$$

$$\therefore\ R_{th_3} = \frac{1}{hA} = \frac{1}{h\pi DL} = \frac{1}{(7270\ \frac{W}{m^2{}^\circ C})\pi(44.45\times10^{-3} m)} = \underline{9.850\times10^{-4}\ \frac{m^\circ C}{W}}$$

▶ R_{th_4} - Convection of Water[1], outer wall

(same as R_{th_3} except for D)

$$R_{th_4} = \frac{1}{(7270\ \frac{W}{m^2{}^\circ C})\pi(52.50\times10^{-3} m)} = \underline{8.340\times10^{-4}\ \frac{m^\circ C}{W}}$$

▶ R_{th_5} - Conduction through Steel

$$R_{th_5} = \frac{\ln\left(\frac{60.33}{52.50}\right)}{2\pi(15.2\ W/mK)} = \underline{1.456\times10^{-3}\ \frac{m^\circ C}{W}}$$

▶ R_{th_6} - Conduction through Intermediate LBE

$K_{LBE}(150^\circ C) = 11.15\ W/mK$

$$R_{th_6} = \frac{\ln\left(\frac{62.71}{60.33}\right)}{2\pi(11.15\ W/mK)} = \underline{5.523\times10^{-4}\ \frac{m^\circ C}{W}}$$

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▶ R_{th_7} – Conduction through Steel

$$R_{th_7} = \frac{\ln\left(\frac{73.03}{62.71}\right)}{2\pi\,(15.2\ W/mK)} = 1.595\times10^{-3}\ \frac{m\,°C}{W}$$

▶ R_{th_8} – Convection of LBE

$$\dot{V} = 1.25\ m^3/hr = 3.47\times10^{-4}\ m^3/s$$

$$A_c = \frac{\pi}{4}(OD^2 - ID^2) = \frac{\pi}{4}\left[(90.12\times10^{-3}m)^2 - (73.03\times10^{-3})^2\right] = 2.19\times10^{-3}\ m^2$$

$$u = \frac{\dot{V}}{A_c} = 0.158\ m/s$$

- LBE properties @ 300°C
  - $\rho = 10364\ kg/m^3$
  - $\nu = 1.87\times10^{-7}\ m^2/s$
  - $k = 12.66\ W/mK$
  - $P_r = 0.0224$
  - $C_p = 146.4\ J/kg°C$

$$D_h = OD - ID = 1.71\times10^{-2}\ m \quad , \quad r^* = \frac{ID}{OD} = 0.81$$

$$Re_{D_h} = \frac{u D_h}{\nu} = \frac{(0.158\ m/s)(1.71\times10^{-2}m)}{(1.87\times10^{-7}\ m^2/s)} = 14448$$

- use Kays & Leung and interpolate
  - A] • uniform wall heat flux
    - $q''_{outer} = 0$
    - thermal entry length negligible (conservative)
    - $r^* \approx 0.8$

P_r	$Re_D = 10^4$ Nu_{ii}	$Re_D = 14448$ Nu_{ii}	$Re_D = 3\times10^4$ Nu_{ii}
0.01	5.95		6.07
0.0224	$\boxed{6.11}$	$\boxed{6.24}$	$\boxed{6.68}$
0.03	6.20		7.05

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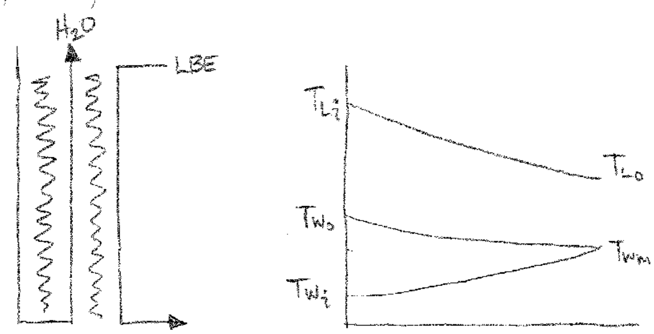
$$h = Nu_D \frac{k}{D_h} = (6.24)\frac{(12.66\ W/mK)}{(1.71\times10^{-2}\ m)} = 4620\ W/m^2{}^\circ C$$

$$\therefore R_{th_s} = \frac{1}{hA} = \frac{1}{h\pi DL} = \frac{1}{(4620\ W/m^2{}^\circ C)\,\pi\,(73.03\times10^{-3}\ m)} = \underline{9.434\times10^{-4}\ \frac{m\,^\circ C}{W}}$$

Heat Transfer Equations

- Q flows from LBE → water① → water②



$$\left.\begin{aligned} Q_L &= \dot{m}_L C_{pL}(T_{Li} - T_{Lo}) \\ Q_W &= \dot{m}_W C_{pW}(T_{Wo} - T_{Wi}) \end{aligned}\right\} Q_L = Q_W$$

$$\left.\begin{aligned} Q_{WW,i} &= \dot{m}_W C_{pW}(T_{Wm} - T_{Wi}) \\ Q_{WW,o} &= \dot{m}_W C_{pW}(T_{Wo} - T_{Wm}) \\ Q_{UALW} &= UA_{LW}\ \frac{(T_{Li} - T_{Wo}) - (T_{Lo} - T_{Wm})}{\ln\left[\frac{(T_{Li} - T_{Wo})}{(T_{Lo} - T_{Wm})}\right]} \\ Q_{UAWW} &= UA_{WW}\left(\tfrac{1}{2}\right)(T_{Wo} - T_{Wi}) \end{aligned}\right\} \begin{aligned} Q_{UAWW} &= Q_{WW,i} \\ Q_{WW,o} &= Q_{UALW} - Q_{UAWW} \end{aligned}$$

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- UA_{LW}

$$\frac{UA}{L} = \frac{1}{R_{th_{tot}}} = \frac{1}{R_{th_4} + R_{th_5} + R_{th_6} + R_{th_7} + R_{th_8}} = 185.8 \frac{W}{m^\circ C}$$

- UA_{WW}

$$\frac{UA}{L} = \frac{1}{R_{th_{tot}}} = \frac{1}{R_{th_1} + R_{th_2} + R_{th_3}} = 196.2 \frac{W}{m^\circ C}$$

- Solution
  - now use MS Excel to solve equations simultaneously

  - Length
    - based on the Russian drawings, the heat transfer length is estimated to be 0.76 m

The results are shown below:

l	0.76	ql	39594
ualw	141.2	qw	39594
uaww	149.1	qualw	39594
cpl	146.6		
cpw	4178	qwwi	852
ml	3.596	qwwo	38741
mw	0.829	quaww	852
tli	350		
twi	25	qualwww	38741
tlo	274.9		
twm	25.2		
two	36.4		

- with the Russian inlet conditions, the heat transfer is 39.6 kW using the present heat transfer coefficients; in order to achieve 52kW, the heat transfer length would have to be increased 39% to 1.06 m.

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▶ Check Kays & Leung against Dwyer

$$Nu = \alpha + \beta (\Psi Pe)^{\gamma} \quad \text{for} \quad 1 < r^* < 7$$

$$\alpha = 4.82 + 0.69 r^*$$
$$\beta = 0.0222$$
$$\gamma = 0.758 r^{*0.053}$$

$$\Psi = 1 - \frac{1.82}{Pr\left(\varepsilon_m/\nu\right)^{1.4}_{max}}$$

$$\left(\frac{\varepsilon_m}{\nu}\right)_{max} = \left(\frac{1}{2}\right)(0.037)\, Re \sqrt{f}$$

$$\frac{1}{\sqrt{f}} = 1.7372 \ln\left[\frac{Re}{1.964 \cdot \ln(Re) - 3.8215}\right]$$

$$\therefore \quad r^* = \frac{90.12}{73.03} = 1.23$$
$$\alpha = 5.67$$
$$\beta = 0.0222$$
$$\gamma = 0.766$$

$$\frac{1}{\sqrt{f}} = 1.7372 \cdot \ln\left[\frac{14448}{(1.964)\ln(14448) - 3.8215}\right] = 11.94$$

$$\left(\frac{\varepsilon_m}{\nu}\right)_{max} = 22.4$$

$$\Psi = 1 - \frac{1.82}{(0.0224)(22.4)^{1.4}} = -0.05 < 0 \quad \therefore \quad \underline{\Psi = 0}$$

$$Pe = Re\, Pr = 324$$

$$Nu = 5.67 + (0.0222)(\overset{0}{\phi} \cdot Pe)^{0.766} \Rightarrow Nu = 5.67$$

- this lower Nu results in a heat transfer rate of 39.1 kW

▶ What happens if Nu is arbitrarily doubled to 12.5?

$\Rightarrow Q = 42.9\,\text{kW}$

1/2

Examine a Range of LBE flow conditions

		$\dot{V}$
1)	Russian flow condition ($1.25\ m^3/hr$)	$3.47\times10^{-4}\ m^3/s$
2)	1.5 m/s in 1" Sch. 40 pipe	$8.36\times10^{-4}\ m^3/s$
3)	1.0 m/s in 2" sch. 40 pipe	$2.16\times10^{-3}\ m^3/s$
4)	2.0 m/s in 2" sch. 40 pipe	$4.33\times10^{-3}\ m^3/s$

• estimate $\overline{T}_{LBE}$ based on previous calcs

$T_{LBE,i}$ →	350°C	450°C	500°C
1)	310°C	400°C	440°C
2)	330°C	420°C	470°C
3)	340°C	440°C	490°C
4)	345°C	445°C	495°C

ave = 410°C

• LBE properties @ 410°C

$\rho = 10230\ kg/m^3$
$\nu = 1.55\times10^{-7}\ m^2/s$
$k = 13.80\ W/m^2\text{°C}$
$Pr = 0.0168$
$C_p = 146.37\ J/kg\text{°C}$

• Reynolds #, Nu_D, h

1) $u = \frac{\dot{V}}{A_c} = \frac{3.47\times10^{-4}\ m^3/s}{2.19\times10^{-3}\ m^2} = 0.158\ m/s,\quad Re_{D_h} = \frac{(0.158)(1.71\times10^{-2})}{(1.55\times10^{-7})} = 17431$

$Nu_D = 6.17 \Rightarrow h = (6.17)\frac{(13.80)}{(1.71\times10^{-2})} = 4979\ W/m^2\text{°C}$

2) ..., $h = 5441\ W/m^2\text{°C}$

3) ..., $h = 6965\ W/m^2\text{°C}$

4) ..., $h = 9636\ W/m^2\text{°C}$

• h using Dwyer
1) $4576\ W/m^2\text{°C}$
2) 6491 "
3) 9743 "
4) 13884 "

2/2

• R_{th_8} - Convection of LBE

$R_{th_8} = \frac{1}{h \pi D L}$, $D = 73.03 \times 10^{-3}$ m

		Dwyer
1)	8.754×10^{-4} $\frac{m°C}{W}$	9.525×10^{-4}
2)	8.011×10^{-4}	6.715×10^{-4}
3)	6.258×10^{-4}	4.474×10^{-4}
4)	4.523×10^{-4}	3.139×10^{-4}

• UA_{LW}

		Dwyer
1)	188.2 W/m°C	185.5
2)	190.8	195.7
3)	197.5	204.7
4)	204.5	210.4

• $\dot{m}$

1) $\dot{m} = \rho \dot{V} = (10230)(3.47 \times 10^{-4}) = 3.55$ kg/s
2) 8.55 kg/s
3) 22.1
4) 44.3

• Q (kW)

				Dwyer		
$T_{LBE,i} \Rightarrow$	350°C	450°C	500°C	350°C	450°C	500°C
1)	40.0	52.3	58.4	39.5	51.6	57.7
2)	43.6	57.1	63.8	44.7	58.4	65.3
3)	46.7	61.0	68.2	48.3	63.2	70.6
4)	48.8	63.8	71.3	50.2	65.6	73.3

Appendix B
Pressure Drop Analysis

1/3

LBE Pressure Drop (Russian Design Point Flow)

▶ for Δp in concentric annulus, use Idelchik diag. 2-7

$$\frac{d}{D} = 0.81 \ , \ Re_{D_h} = 1.4\times10^4 \Rightarrow K_{nonc} \approx 1.07$$

$$\lambda = \frac{0.3164}{Re^{0.25}} = \frac{0.3164}{(14448)^{0.25}} = 0.0289$$

$$f = K_{nonc}\,\lambda = 0.0309$$

$$\Delta p = \frac{fL}{D}\left(\tfrac{1}{2}\rho u^2\right) = \frac{(0.0308)(0.76m)}{(1.71\times10^{-2}m)}\left(\tfrac{1}{2}\right)(10364\ kg/m^3)(0.158\ m/s)^2$$

$$\underline{\Delta p = 177\ Pa}$$

▶ use Idelchik diag. 4-9 (beveled) for exit

- annulus area, $A_A = 2.19\times10^{-3}\,m^2$
- exit area, $A_E = \frac{\pi}{4}(5.25\times10^{-2}m)^2 = 2.16\times10^{-3}\,m^2$

} negligible!

▶ @ 2-90° elbows, use Idelchik diag. 6-1 (@ 1¼" ID)

$$K = K_{loc} + 0.0175\,\delta\lambda\frac{R_o}{D_h}$$

$$\left.\begin{array}{l} K_{loc} = A\cdot B\cdot C \\ A = 1.0 \\ B = 1.19 \ (@\ R/D = 0.5) \\ C = 1 \end{array}\right\} K_{loc} = 1.19$$

$$K_{fr} = 0.0175(90)(0.0289)(0.5) = 0.023$$

$$K = (1.19 + 0.023) = 1.213$$

$$A_c = 7.92\times10^{-4}\,m^2 \Rightarrow u = 0.44\,m/s$$

2 elbows

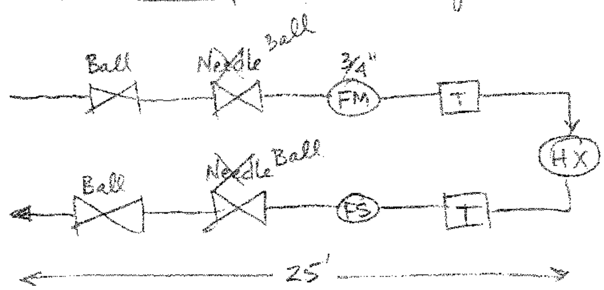
$$\Delta p = (2)(1.213)\left(\tfrac{1}{2}\right)(10364)(0.44)^2 = \underline{2434\ Pa}$$

2/3

▷ Δp on LBE side

$$\Delta p = 177 + 2434 = 2611 \text{ Pa} \cong \underline{0.4 \text{ psi}}$$

<u>Water Pressure Drop</u> (Russian Design Point Flow)



<u>A</u>| 3/4"

- ball valve: $C_V = 31$, orifice = 22.2 mm
- needle valve: $C_V = 1.8$, orifice = 9.5 mm
- FM (FS): 3.6 psi @ 11 gpm

▷ Ball Valve

$$\left.\begin{aligned} \Delta p &= 1 \text{ psi} = 6895 \text{ Pa} \\ \dot{V} &= 31 \text{ gpm} = 1.96 \times 10^{-3} \text{ m}^3/\text{s} \\ A_c &= 3.87 \times 10^{-4} \text{ m}^2 \\ V &= 5.05 \text{ m/s} \end{aligned}\right\} \quad k_{BV} = \frac{6895}{\frac{1}{2}(1000)(5.05)^2} = 0.54$$

▷ Needle Valve

$$\left.\begin{aligned} \Delta p &= 6895 \text{ Pa} \\ \dot{V} &= 1.8 \text{ gpm} = 1.14 \times 10^{-4} \text{ m}^3/\text{s} \\ A_c &= 7.09 \times 10^{-5} \text{ m}^2 \\ V &= 1.6 \text{ m/s} \end{aligned}\right\} \quad k_{NV} = \frac{6895}{\frac{1}{2}(1000)(1.6)^2} = 5.4$$

▷ Flow Meter

$$\left.\begin{aligned} \Delta p &= 3.6 \text{ psi} = 24820 \text{ Pa} \\ \dot{V} &= 11 \text{ gpm} = 6.94 \times 10^{-4} \text{ m}^3/\text{s} \\ \underline{A}| \ A_c &= 2.85 \times 10^{-4} \text{ m}^2 \\ V &= 2.44 \text{ m/s} \end{aligned}\right\} \quad K_{FM} = \frac{24820}{\frac{1}{2}(1000)(2.44)^2} = 8.34$$

3/3

▶ Tubing

$$K_t = \frac{fL}{D} \quad \underline{Al}\ f = 0.03,\ L = 15.2\,m,\ D = 0.0191\,m$$

$$K_t = \frac{(0.03)(15.2)}{(0.0191)} = 23.9$$

▶ fittings

$\underline{Al}$ 10 threaded 90° elbows @ $K_{el} = 1.5$ ea

1, line flow, threaded Tee @ $K_T = 0.9$

▶ HX

• annulus – similar f to LBE

$$K_a = \frac{fL}{D} = \frac{(0.0309)(1.5)}{8.05\times10^{-3}} = 5.8$$

• inner tube

$$K_b = \frac{fL}{D} = \frac{(0.03)(1.5)}{4.02\times10^{-2}} = 1.1$$

▶ total Δp

• $A_{ref} = 2.85\times10^{-4}\,m^2 \quad (D = \tfrac{3}{4}'')$, $\quad V_{ref} = \frac{\dot{V}}{A_{ref}} = \frac{8.33\times10^{-4}\,m^3/s}{2.85\times10^{-4}\,m^2} = 2.9\,m/s$

$$\Delta p = \Sigma k \cdot \tfrac{1}{2}\rho V_{ref}^2 = A_{ref}^2\left(\frac{4K_{BV}}{A_{BV}^2} + \cancel{\frac{2K_{NV}}{A_{NV}^2}} + \frac{2K_{FM}}{A_{FM}^2} + \frac{K_t}{A_t^2} + \frac{10K_{el}}{A_{el}^2} + \frac{K_T}{A_T^2} + \frac{K_a}{A_a^2} + \frac{K_b}{A_b^2}\right)\tfrac{1}{2}\rho V_{ref}^2$$

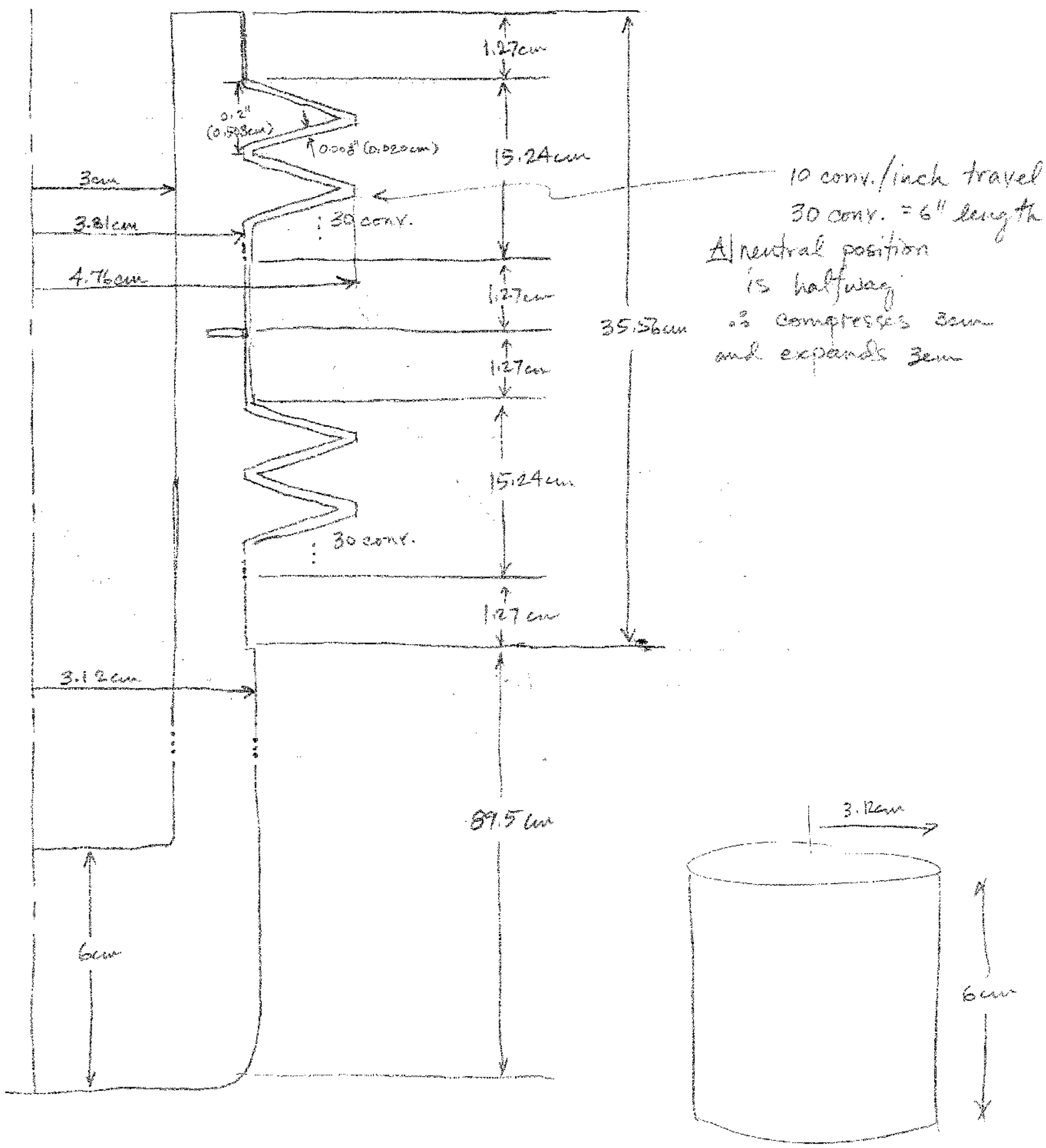
$$\Delta p = (2.85\times10^{-4})^2\left[\frac{(4)(0.54)}{(3.87\times10^{-4})^2} + \cancel{\frac{(2)(5.4)}{(7.69\times10^{-5})^2}} + \frac{(2)(8.34)}{(2.85\times10^{-4})^2} + \frac{(23.9)}{(2.85\times10^{-4})^2} + \frac{(10)(1.5)}{(2.85\times10^{-4})^2}\right.$$

$$\left. + \frac{(0.9)}{(2.85\times10^{-4})^2} + \frac{(5.8)}{(6.13\times10^{-4})^2} + \frac{(1.1)}{(1.27\times10^{-3})^2}\right]\left(\tfrac{1}{2}\right)(1000)(2.9)^2$$

$$\Delta p = (59.0)\left(\tfrac{1}{2}\right)(1000)(2.9)^2 = 248\,kPa = \underline{36.0\ psi}$$

Appendix C
Internal Pressure Analysis

11/3/00 - Perform Analysis of Δp in HX Intermediate Section 1/4



$V_{LBE} = \pi r^2 L = 183.3\ cm^3$

2/4

Compressed Volume, Empty

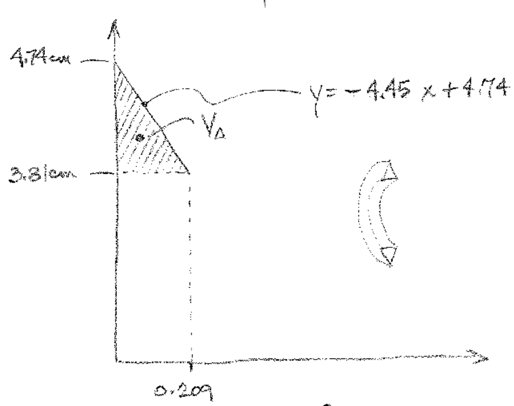
1) LBE compartment + annulus

$$V_1 = \pi(r_o^2 - r_i^2)L = \pi(3.12\,cm^2 - 3\,cm^2)(89.5\,cm) = \underline{206.5\,cm^3}$$

2) Inner bellows compressed annulus + bellows attachments

$$V_2 = \pi(r_o^2 - r_i^2)L = \pi(3.81\,cm^2 - 3\,cm^2)(32.6\,cm) = \underline{564.9\,cm^3}$$

3) Bellows, compressed



$y = mx + b$
$b = 4.74$

$$m = \frac{(3.81 - 4.74)}{0.209 - 0} = -4.45$$

$$\frac{1}{2} V_\Delta = \pi \int_0^{0.209} \left[(-4.45x + 4.74)^2 - 3.81^2\right] dx$$

$$= \pi \int_0^{0.209} (19.80x^2 - 42.19x + 7.95)\,dx$$

$$= \pi \left[19.80\frac{x^3}{3} - 42.19\frac{x^2}{2} + 7.95x\right]_0^{0.209}$$

$$= \pi(0.060 - 0.921 + 1.66) = 2.51\,cm^3$$

$$V_\Delta = 2(2.51) = 5.02\,cm^3$$

$$V_3 = 60\,V_\Delta = \underline{301.2\,cm^3}$$

$$\therefore V_{comp} = V_1 + V_2 + V_3 = \underline{1073\,cm^3}$$

3/4

Expanded Volume, Empty

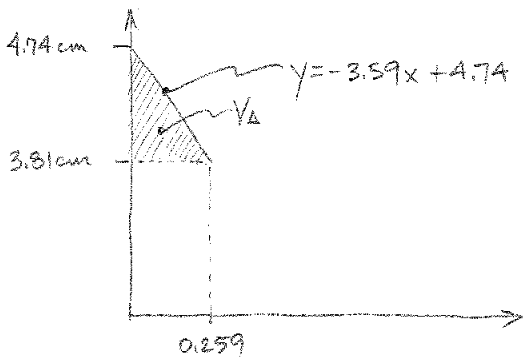
1) LBE compartment + annulus

$$V_1 = 183.3\,cm^3 + \pi\left(3.12\,cm^2 - 3\,cm^2\right)(83.5\,cm) = \underline{376.0\,cm^3}$$

2) Inner bellows expanded annulus + attachments

$$V_2 = \pi\left(3.81\,cm^2 - 3\,cm^2\right)(38.6\,cm) = \underline{668.9\,cm^3}$$

3) Bellows, expanded



$y = mx + b$

$b = 4.74$

$$m = \frac{3.81 - 4.74}{0.259 - 0} = -3.59$$

$$\frac{1}{2} V_\Delta = \pi \int_0^{0.259} \left[(-3.59x + 4.74)^2 - 3.81^2 \right] dx$$

$$= \pi \int_0^{0.259} \left(12.89x^2 - 34.03x + 7.95\right) dx$$

$$= \pi \left[12.89\frac{x^3}{3} - 34.03\frac{x^2}{2} + 7.95x \right]_0^{0.259}$$

$$= \pi\left(0.0747 - 1.141 + 2.059\right) = 3.117\,cm^3$$

$$V_\Delta = 2(3.117) = 6.23\,cm^3$$

$$V_3 = 60\,V_\Delta = \underline{374.1\,cm^3}$$

$$\therefore\ V_{exp} = V_1 + V_2 + V_3 = \underline{\underline{1419\,cm^3}}$$

4/4

▶ Volume Ratio

$$V_C = V_{comp} - V_{LBE} = 1073\ cm^3 - 183\ cm^3 = 890\ cm^3$$

$$V_E = V_{exp} - V_{LBE} = 1419 - 183 = 1236\ cm^3$$

$$\frac{V_E}{V_C} = \frac{1236}{890} = 1.39$$

▶ Temperature Ratio

$$T_{fill} = 293K$$

$$T_{max} = 500°C = 773\ K$$

$$\frac{T_{max}}{T_{fill}} = \frac{773}{293} = 2.64$$

▶ Pressure Rise

$$PV = nRT \Rightarrow \frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \Rightarrow \frac{P_2}{P_1} = \frac{V_1}{V_2}\frac{T_2}{T_1}$$

$$\frac{P_2}{P_1} = (1.39)(2.64) = 3.67$$

$$\text{say } SF = 1.5 \Rightarrow \left.\frac{P_2}{P_1}\right|_{SF} = (1.5)(3.67) = 5.5$$

$$\text{if } p_1 = 1 atm, \quad p_2 = 3.67 atm\ (54\ psi)$$

$$p_{s_{SF}} = 5.5\ atm\ (81\ psi)$$